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by

R. A. Yanson

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Б б	*Б б*	B, b	С с	*С с*	S, s
В в	*В в*	V, v	Т т	*Т т*	T, t
Г г	*Г г*	G, g	У у	*У у*	U, u
Д д	*Д д*	D, d	Ф ф	*Ф ф*	F, f
Е е	*Е е*	Ye, ye; E, e*	Х х	*Х х*	Kh, kh
Ж ж	*Ж ж*	Zh, zh	Ц ц	*Ц ц*	Ts, ts
З з	*З з*	Z, z	Ч ч	*Ч ч*	Ch, ch
И и	*И и*	I, i	Ш ш	*Ш ш*	Sh, sh
Й й	*Й й*	Y, y	Щ щ	*Щ щ*	Shch, shch
К к	*К к*	K, k	Ъ ъ	*Ъ ъ*	"
Л л	*Л л*	L, l	Ы ы	*Ы ы*	Y, y
М м	*М м*	M, m	Ь ь	*Ь ь*	'
Н н	*Н н*	N, n	Э э	*Э э*	E, e
О о	*О о*	O, o	Ю ю	*Ю ю*	Yu, yu
П п	*П п*	P, p	Я я	*Я я*	Ya, ya

*ye initially, after vowels, and after ъ, ь; e elsewhere. When written as ё in Russian, transliterate as yё or ё. The use of diacritical marks is preferred, but such marks may be omitted when expediency dictates.

GREEK ALPHABET

Alpha	Α	α ∝	Nu	Ν	ν
Beta	Β	β	Xi	Ξ	ξ
Gamma	Γ	γ	Omicron	Ο	ο
Delta	Δ	δ	Pi	Π	π
Epsilon	Ε	ε ϵ	Rho	Ρ	ρ ϱ
Zeta	Ζ	ζ	Sigma	Σ	σ ς
Eta	Η	η	Tau	Τ	τ
Theta	Θ	θ ϑ	Upsilon	Τ	υ
Iota	Ι	ι	Phi	Φ	φ ϕ
Kappa	Κ	ϰ κ	Chi	Χ	χ
Lambda	Λ	λ	Psi	Ψ	ψ
Mu	Μ	μ	Omega	Ω	ω

RUSSIAN AND ENGLISH TRIGONOMETRIC FUNCTIONS

Russian	English
sin	sin
cos	cos
tg	tan
ctg	cot
sec	sec
cosec	csc
sh	sinh
ch	cosh
th	tanh
cth	coth
sch	sech
csch	csch
arc sin	$\sin^{-1}$
arc cos	$\cos^{-1}$
arc tg	$\tan^{-1}$
arc ctg	$\cot^{-1}$
arc sec	$\sec^{-1}$
arc cosec	$\csc^{-1}$
arc sh	$\sinh^{-1}$
arc ch	$\cosh^{-1}$
arc th	$\tanh^{-1}$
arc cth	$\coth^{-1}$
arc sch	sech^{-1}
arc csch	csch^{-1}

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rot	curl
lg	log

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STUDY ON STAGE OF CENTRIFUGAL COMPRESSOR WITH AIR TURBINE

Assistant R. A. Yanson

(Article presented by Doctor of Technical Sciences V. V. Uvarov, Professor at Moscow Higher Technical School im. N. E. Bauman)

ABSTRACT Discussed briefly in this article is the scheme of a centrifugal compressor with an air turbine, which replaces the bladed diffuser. A brief description is given of the experimental device and the experimental characteristics of this stage. Investigated is the problem of optimizing parameters in the given compressor. END ABSTRACT

To achieve the compression process in a gas-turbine engine (GTE) in the case where relatively small volumetric air flows pass through it, a centrifugal compressor is generally used, whose efficiency seldom exceeds 0.8. This value can be somewhat increased and a number of new GTE properties obtained when a centrifugal compressor with an air turbine is used in place of the bladed diffuser. Such a compressor (henceforth it will be called a turbine compressor) was first proposed by Professor V. V. Uvarov*

[FOOTNOTE: Inventor's certificate No. 27155, class 27.9, 7 March 1931. END FOOTNOTE]

Usually the turbine compressor stage has a configuration close to that shown in Fig. 1. The bladed diffuser, which is rigidly connected to the housing, is replaced by a centrifugal radial air turbine mounted on independent bearings and rotating in the same direction as the impeller. Thus, the centrifugal wheel is located inside the drum rotor of the turbine. Air passes through inlet stator 1, rotating inlet blade row 2, axisymmetrical bend 3, impeller 4, and then slotted diffuser 5. As it passes through the channels of air

turbine 6, the energy of the air, which is primarily speed energy, is transformed into mechanical work. Through the rotating inlet blade row this power is transmitted to the rotor and then to the consumer. Thus, the work which is transmitted to the rotor is spent on compression of the air and rotation of the shaft of the air turbine. This means that in a gas-turbine engine with a turbine compressor the power turbine is moved from the gas portion of the tract to the air portion and the turbine compressor represents a kind of pneumatic reducer, since the number of revolutions of the air turbine is usually about one third the number of revolutions of the centrifugal wheel. The presence of a rotating housing around the impeller and the rotation of the walls of the slotted diffuser increases the efficiency of the compression process in these parts.

The problem of compression for small volumes of air arises also in high-powered gas-turbine installations (200 thousand kW or above) in the last high-pressure compressor. In this case the so called axial turbine compressor [1] can be used. It is a two-stage compressor: The first stage is the axial compressor, the second - the turbine. The consumer of all of the power of the air turbine in this case is the axial compressor, whose compressed air goes to the turbine compressor.

The turbine compressor has recently been studied in Department

E-3 at MVTU [Higher Technical School im. N. E. Bauman]. For this purpose a special stand consisting of a drive (internal combustion engine), two boosters, a experimental turbine compressor, a hydraulic brake to absorb the power of the air turbine, maintenance systems, measuring systems, and a control panel, was designed, built, and installed.

The most complete results were obtained in testing a compressor designed for an air flow rate of 1.1 kg/s. The compressor had a cantilever arrangement of the rotors of the impeller and turbine. The inlet stator consisted of a circular cascade of 36 sheet-type vanes, whose inlet sections could be turned for the purpose of measuring the twist in the flow in front of the impeller. The rotating inlet blade row was made in the form of 12 cylindrical cross bars measuring 7 mm in diameter to simplify analysis of its losses in variable modes. The impeller had 24 radial blades; diameter ratio D_1/D_2 was 0.57 for an outer diameter of D_2 = 293 mm. The slotted diffuser and the air turbine had a constant width of 17.2 mm. The diameter ratio $D_2/D_3 = \zeta_2 = 0.9$, while $D_4/D_3 = \zeta_4 = 1.1$. The channels of the air turbine were slightly convergent. The measuring sections in the flow-through part were located in front of the inlet stator and in the slotted diffuser behind the air turbines.

The stand systems were designed for measuring the necessary

parameters. The air flow G_0 through the inlet section of the compressor OO (Fig. 1) was measured by means of a standard meter ring. To determine the flow of air G_1 passing through the impeller a correction (of about 3-4o/o) for leakage of air through labyrinths L_1 and L_2 (Fig. 1) was introduced. Pressure was measured by means of vents and combs (including those which could be rotated by remote control) and readings were issued on group recording manometers. The temperature was established by chromel-copel thermocouples. Readings were issued on an electronic recording potentiometer. The number of revolutions was determined by standard ferromagnetic stand tachometers. The torque transmitted to the rotor of the impeller was measured by a reducer-balancer, friction losses in which were considered by calibration, while the torque on the turbine rotor was measured by a hydrodynamometer and the housing of the hydromotor. Friction losses in the bearings were determined by a special system through heat transfer to the oil. To obtain stable results no less than four measurements were taken in each mode.

From the results of the measurements the sum of parameters describing the stage were calculated:

total pressure ratio $\pi_k^* = \frac{p_4^*}{p_0^*}$;

adiabatic work of compression $H_{ad} = \frac{k}{k-1} RT_0^* \left(\pi_k^{*\frac{k-1}{k}} - 1\right)$;

power conducted to impeller and spent on compression of air in it $N^{кр}_{сж}$;

power developed on rim of air turbine $N^{вт}_{лоп}$;

power spent on compression of air in the stage, $N_{сж} \simeq N^{кр}_{сж} - N^{вт}_{лоп}$;

coefficient of power transfer $\chi = \frac{N^{вт}_{лоп}}{N^{кр}_{сж}}$;

efficiency of stage $\eta^{*}_{ад} = \frac{G_0 H_{ад}}{N_{сж}}$.

All quantities were reduced to normal atmospheric conditions.

The compressor characteristics were recorded on the following impeller revolutions $n_{кр}$: 14,000, 16,000, 17,800 r/min with relative revolutions of the air turbine of $\bar{n} = \frac{n_{кр}}{n_{вт}} = 0;\ 0,2;\ 0,3;\ 0,4;\ 0,45$.

Figure 2 shows the partial results of tests for a constant number of impeller revolutions and for constant relative revolutions of the turbine $\bar{n}$. We first see that for the calculated air flow rate the coefficient of the adiabatic pressure head of the stage $\bar{H}_{ад} = 0,45 \div 0,52$. The decrease in this quantity as compared to its value in

a standard centrifugal compressor stage results from the fact that not all of the power conducted through the impeller went to the compression of air. The fraction of power X transmitted to the shaft of the air turbine, as we see in Fig. 2, is greatly dependent on the mode, and in the region of maximal efficiency of the stage constitutes about a third of the power conducted to the impeller. When the air turbine is damped, the coefficient of the adiabatic pressure head rises substantially. Here there is a simultaneous increase in the torque on the turbine shaft (more than twofold, as shown by the experiment). As the number of revolutions of the air turbine grows, all of the characteristics shifted toward lower air flow rates. The maximal efficiency for the given impeller revolutions for relative revolutions $\bar{n}=0{,}4$ was 0.82. For lower numbers of impeller revolutions an efficiency of 0.84 was reached. The sharp drop in the intensity of surging pulsations in the rotation of turbine should also be noted along with the fact that the ratio of the maximal flow through the compressor to the flow corresponding to the beginning of surging pulsations increased from 1.7 to 2.7 as the number of turbine revolutions increased from $\bar{n}=0$ to 0.3 for constant numbers of impeller revolutions in $n_{кр}$ = 16,000 r/min.

Theoretical analysis makes it possible to establish the existence of an optimal interrelationship between the design and gasdynamic parameters for an isolated turbine compressor stage. For

this purpose a method of calculating the stage was developed on the basis of [2] with careful differentiation of losses and various types of friction work [3]. This was based on existing experimental material obtained from studying the parts of centrifugal compressors and impulse turbines. Calculations were conducted on the "Ural-2" digital computer. The stage geometry was assumed to be that of Fig. 1, the impeller blades - radial, the channels of the air turbine - slightly convergent, and the intake parameters - atmospheric.

As an example Fig. 3 shows some calculation results for the following fixed quantities: diameter ratio $D_1/D_2 = 0.65$, ratio of number of revolutions $\bar{n}=0{,}3$, coefficient of flow rate at wheel inlet $\bar{c}_{a\,cp}=\frac{c_{a\,cp}}{u_1}=0{,}3$, ratio of velocity circulations at inlet and outlet from wheel $\frac{\pi D_{1\,cp} c_{1u\,cp}}{\pi D_2 c_{2u}}=0{,}1$, peripheral velocity on outer diameter of wheel u_2 = 350 m/s. In particular these curves enable us to establish the optimal angle at which the flow emerges from the channels of the wheel (impeller) in absolute motion α_2 for a fixed quantity of work taken from the shaft of the air turbine $\bar{L}_{в\,т}=\frac{L_{в\,т}}{u_2^2/g}$. It should be noted that the pressure ratio in the stage π_k^* for a fixed quantity of work $\bar{L}_{в\,т}$ changes only slightly with a change in α_2, while the angle at which the flow emerges from the channels of the gas turbine in absolute motion α_4 changes substantially. In order to achieve a radial discharge of the flow from the turbine, each value of the work $\bar{L}_{в\,т}$ requires its own radial length of slotted diffuser ζ_2.

When we move to higher velocities u_2 the general nature of the curve remains as before, although the maximally attainable efficiency level of the stage η^*_{ad} drops. When the ratio of the number of revolutions $\bar{n}$ is changed in a range of from 0.2 to 0.4, we observe a continuous, although slow, rise in efficiency, which is also illustrated by Fig. 3. Thus, the optimal ratio of revolutions $\bar{n}$ found from the condition of maximal efficiency of the entire compressor does not coincide with the optimal ratio $\bar{n}$ found from the condition of maximum efficiency of the air turbine [3]. The latter is as a rule greater than the former (0.25-0.35). Here we should mention that in the given calculations the ratio of the height of the blade of the air turbine to its cord was assumed equal to 1.2, and thus a different radial length was obtained for the rim of the turbine.

By using such curves a designer can select the best parameters for calculating the stage with the quantities assigned to him: pressure ratio in stage π_k^*, work on turbine shaft $L_{s.t}$, stage efficiency η^*_{ad}, air flow G_0 (or flow rate coefficient $\bar{c}_{a\,cp}$, if the wheel geometry and velocity u_2 are fixed).

Conclusion

1. In the case where relatively small volumetric air flows pass through a gas turbine engine (1-2 kg/s), the centrifugal compressor is replaced by a turbine compressor, thus assuring a number of new properties for the engine as a whole for a compression process efficiency of 0.80-0.82.

2. The tests which were conducted on the turbine compressor stage showed that when the flow rate of air was about 1 kg/s, impeller revolutions 16,000 r/min, and relative revolutions of air turbine $\bar{n}=0.4$, efficiency reached 0.82, where about 30o/o of the power conducted to the impeller was transmitted to the output shaft of the turbine.

3. For a turbine compressor stage there exists an optimal relationship between design and gasdynamic parameters, which must be considered in calculation.

Article received 2 March 1966

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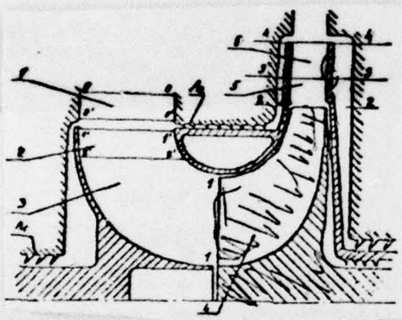
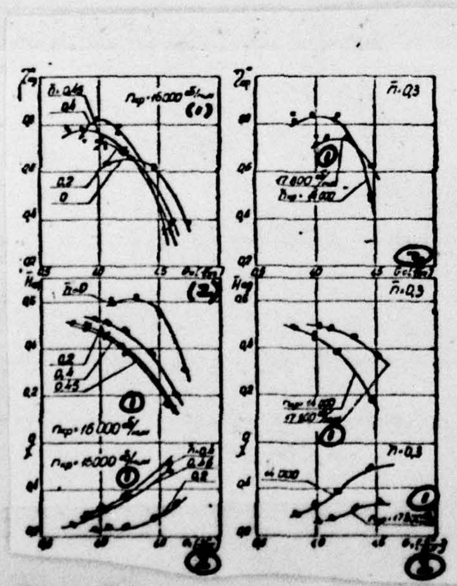


Fig. 1.



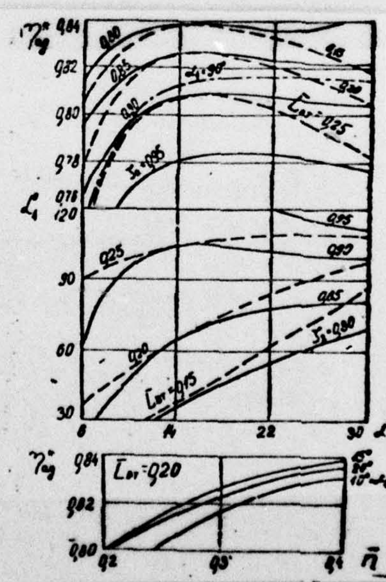


Fig. 2.
Key: (1) r/min, (2) kg/s.

Fig. 3.

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